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Abstract

Results of two different nonlinear finite-element analyses and preliminary static test results for the final design of the Controls-Structures Interaction Evolutionary Model reflector are presented. Load-deflection data bases are generated from analysis and testing of the 16-ft diameter, dish-shaped reflector, and natural frequencies and mode shapes are obtained from vibrational analysis. Experimental and analytical results show similar trends; however, future test hardware modifications and finite-element model refinement would be necessary to obtain better correlation. The two nonlinear analysis approaches are both adequate techniques for the analysis of prestressed structures with complex geometry.

Introduction

Future space structures, such as the proposed *Space Station Freedom*—which consists of a truss structure with many appendages such as antennas and motors—present new challenges to structure and control-system design. The structural design requirement of low mass results in very flexible structures. To be able to meet pointing-control requirements in space, engineers need complete knowledge of the static and dynamic characteristics of the structure.

New technology for ground testing and analysis to characterize controlled flexible space structures is being developed and tested as described in references 1 and 2. Correlation of experimental and analytical results leads to the refinement of the analytical models, which gives engineers more confidence in the analytical predictions. The final goal is to be able to characterize and design space structures by means of analysis only or by means of analysis and testing of individual components of the structure.

Langley Research Center recently conducted closed-loop-control ground tests on the Controls-Structures Interaction Evolutionary Model (CEM), an experimental model that is generically similar to a future space platform to be instrumented to monitor the Earth's climate. Figure 1 shows the main components of the CEM. Preliminary design, test, and analysis results are described in reference 2. As shown in the figure, the Evolutionary Model consists primarily of a flexible truss structure and an antenna-like appendage called a reflector. The reflector, shown in detail in figure 2, is an important dynamic component of the global line-of-sight (LOS) pointing path. To monitor the LOS pointing accuracy, a laser is mounted on the vertical truss tower of the CEM, such that the laser beam reflects upon the reflector mirror. The laser-beam reflection is measured by a photodiode array above the reflector. This laser-reflector-detector system allows the pointing accuracy of the CEM to be measured and controlled. Because of the complexity of the geometry of the reflector, and in an effort to update the finite-element analytical model of the whole structure, testing and analysis of that individual component have been conducted. Reference 3 presents preliminary design, test, and analysis results of the developmental model of the reflector. The present paper describes the results obtained from the finite-element analysis and static test for the final design of the reflector and some preliminary results from vibrational analysis. Nonlinear capabilities of MSC/NASTRAN (ref. 4) were used to account for large-displacements and pretensioning effects in the finite-element analysis of the reflector; results were compared with the nonlinear technique described in reference 3.

Evolutionary Model Reflector

The CEM reflector (figs. 2 and 3) is a dish-shaped structure 185.5 in. in diameter and 19.93 in. deep. The main components are the ribs, hub, and sensor plate. Each of the eight aluminum ribs is 0.25 in. thick and 96 in. long and is tapered in width over its length from 2 in. to 1 in. The ribs are oriented at angles of 45° around the hub—a 3/8-in.-thick aluminum plate, with a 4-in. inside diameter and an 8-in. outside diameter. One end of the ribs is attached to the hub, and the other end is connected to each adjacent rib by a 1/32-in.-diameter steel cable. Tensioning the cable by means of thumb screws on each rib deforms the ribs to obtain the desired shape of the reflector.

The sensor plate is a 1.5-in.-thick fiberglass-honeycomb composite panel with a mirrored surface. The top view of the reflector in figure 3 reveals the octagonal shape of the reflector plate and the circular mirror on its center. Each corner of the octagonal panel is attached to the ribs by swivel-head bolts to prevent transmission of moments from the ribs to the panel. A detailed view of that connection is shown in figure 3. Four aluminum rods stiffen the plate and connect it to the hub. The hub is the connecting linkage between the reflector and the supporting structure. A detailed view of the connections between the hub and sensor plate and between the hub and truss tower is shown in figure 4.

During this investigation, the reflector was statically tested in two positions—horizontally (fig. 5) and inclined 39.1° (fig. 6). The inclined position is the same as for the CEM. It was supported in the horizontal position by a single 10-in. cubical truss bay fixed at the bottom (fig. 5). The supporting structure for the inclined reflector test setup (fig. 6) was the upper section of the truss tower; this tower consisted of a tapered truss bay and one cubic bay that was also fixed at its bottom. The truss members of the cubical bays are aluminum tubes connected by node-ball joints. A typical truss member and node-ball joint are shown in figure 7. The vertical members of the tapered bay are aluminum tubes, and the diagonal and top members are aluminum structural angles. Dynamic analyses were performed only in the inclined position.

Finite-Element Models

The dish shape of the reflector is a result of the deflection of the ribs caused by tensioning the cables. Previous finite-element analysis of a preliminary reflector design (ref. 3) showed that small-deflection nonlinear analysis can be used if the post-tensioned geometry and compressive loads of a typical rib are known. A model of a prestressed reflector following this approach was created by using the MacNeal-Schwendler Corp. MSC/NASTRAN. A second nonlinear analysis, which included MSC/NASTRAN nonlinear analysis capabilities, was used to model the large deflections of the reflector, starting from its undeformed position, to obtain the correct geometry and stiffness of the prestressed structure. The only physical parameter needed for the analysis in this case, other than material properties and basic dimensions, is the tension in the cables for the final configuration. Results from both analyses were compared with test results.

In the finite-element models of the reflector, each rib consists of 12 beam elements dimensioned according to the tapered shape of the ribs. The cables are modeled by using 1/32-in.-diameter rod elements with material properties of steel wire. The hub is modeled with 24 3/8-in.-thick triangular plate elements. The steel bolts connecting the ribs to the hub are represented by 1/4-in.-diameter bar elements. Due to the short length and high stiffness of the bolts connecting the hub to the supporting structure, zero-length scalar spring elements (1.5×10^8 lb/in.) for all six degrees of freedom are used for each connector. All support brackets and truss elements of the supporting structure were modeled by using two-noded CBAR elements.

The sensor plate is modeled by using 24 triangular plate elements. Since the material properties of the honeycomb composite panel were unknown, an effective plate thickness of 0.408 in. was computed, and the known material properties of the fiberglass sheets were used as material properties for the equivalent plate. The following equation was used to compute the effective thickness t_{eff} of the composite panel:

$$I = \frac{t_{\text{eff}}^3}{12} \times b = \frac{b\left(h_o^3 - h_i^3\right)}{12}$$

Therefore,

$$t_{\text{eff}}^3 = \left(h_o^3 - h_i^3\right)$$

where I is the area moment of inertia for a rectangular cross-sectional element of the panel of length b and height h_o. (See fig. 8.) Honeycomb core thickness is denoted by h_i. The mirrored surface of the reflector plate was represented by a lumped mass at its center. The swivel-head bolts connecting the sensor plate to the ribs were modeled with CBAR elements, and the rotational degree of freedom about the axis passing through the eye of each bolt (see detail in fig. 3) was left free by using pin flags. Since CROD elements only have torsional and axial stiffness, they were also used to model the swivel-head bolts; results were compared with those obtained with CBAR elements.

The input geometry of the undeformed rib for the large-displacements nonlinear model should not be represented by a horizontal line. A bifurcation would exist and the ribs could deflect either up or down. To ensure that the ribs would move in the correct direction, the rib was represented by a straight line that made a 6° angle with a horizontal line (fig. 9).

Analysis

The MSC/NASTRAN solution 64 employs an iterative procedure with a modified Newton-Raphson approach to solve geometric nonlinear problems. The large-displacements nonlinear analysis for the reflector involved two steps, which are summarized in figure 10. In the first step, the structure was preloaded and shaped by applying a thermal load to the cables that was equivalent to the measured tension in the cables on the shaped structure. Gravity effects and target weights were also included. Fifteen iterations were required for force convergence, and the first iteration was the linear static solution. Differential stiffness calculations were skipped to avoid instability or mechanism errors. The second step was a restart from step 1 to apply external loads. Fifteen dummy subcases were required in the case control deck to restart from the last stress state in step 1. Three iterations were required for final convergence in step 2. Superimposing results from steps 1 and 2 gives the displacements that result from external loading. These results are compared with small-displacements nonlinear analysis and experimental results.

Analysis with a prestressed reflector model, similar to the analysis described in reference 3, was also performed by using solution 64; however, the geometry input for the ribs was that of a deflected and prestressed rib. Since there were no large deflections of the preshaped structure, the CBEAM elements were replaced by the easier to use CBAR elements. The analysis consisted of the three steps shown in figure 11. First, a thermal load equivalent to the compressive preload is applied to the ribs, which are completely restrained (ref. 3). A thermal preload is also applied to the cables. The constraint forces obtained in this step are the forces required to maintain equilibrium when all degrees of freedom are released in step 2. The second step is to release all degrees of freedom, apply the computed constraint forces, gravity load, and target weights to obtain the final prestress state, which is equivalent to step 1 for the large-displacements nonlinear model. Step 3 involves the application of external loads. Results from steps 2 and 3 are combined to obtain the final displacements. For this case, each step ran independently, no data base was required. Each step required three iterations for convergence—a linear static solution, a differential stiffness calculation, and one nonlinear iteration. Figure 9 shows the geometry of a preloaded rib that results from small-displacements nonlinear analysis and large-displacements nonlinear analyses. Listings of the NASTRAN data decks for both models are included in the appendix.

The analysis results seem very sensitive to different models of swivel-head bolts. Changing the swivel bolt element from CBAR with pin flags to CROD greatly reduces the stiffness of the ribs and smooths the stress distribution along the ribs. Figure 12 shows the deformation of one of the ribs under gravity and target weight for the small-displacements analysis with two different connector models. Significant changes occur in the axial-force distribution along the ribs for the large-displacements nonlinear model. (See table 1.)

Vibrational analysis was also performed by using the data bases generated for the final prestressed states for both the small-displacements and the large-displacements nonlinear analytical models of the reflector in its inclined position. Mode shapes and frequencies were computed for modes below 10 Hz.

Correlation of Static Tests With Analysis

Static tests of the reflector on its horizontal and inclined configurations were conducted to obtain load-deflection characteristics for comparison with analytical results. Four of the eight reflector ribs, numbered as shown in figure 3, were instrumented with target plates and proximity probes to measure rib-tip and plate-end displacements. Loads were applied at specific locations on the ribs and plate ends to provide the required symmetric or unsymmetric loading condition. Loads were applied and removed in step increments. Table 2 summarizes the loading cycles that were conducted to obtain the data base for this investigation; figure 13 shows the details of the target and weight configurations. Output data from the proximity probes were displayed on voltmeters and were recorded manually.

Load-deflection plots for each loading condition described in table 2 were generated from the test data for comparison with load-deflection plots generated from large-displacements nonlinear and small-displacements nonlinear analyses. Symmetric and asymmetric stiffness characteristics of the reflector ribs for test and analysis of the reflector on its inclined position are shown in figure 14. Both sets of data indicate that the load deflections are linear during load-application and load-relief cycles; there is good correlation between small-displacements and large-displacements nonlinear analysis results. As explained subsequently in this section, correlation between experimental and analytical results is acceptable, considering possible errors in experimental measurements. Similar plots were generated that described load-deflection characteristics of the reflector in its horizontal position when loads were applied at the sensor-plate ends. Experimental and analytical results obtained from symmetric and asymmetric loading of the plate ends are shown in figure 15. For this set of data, because of the symmetry of the structure, all the measured and generated displacement data obtained for each of the four locations on the sensor plate were combined and curve fitted. Experimental data show hysteresis losses during the loading and unloading cycles; however, load-deflection characteristics can be considered linear. Hysteresis loss is a common characteristic of composite material structures. Even though the present analytical tools do not have the capabilities to model hysteretic energy losses, load-deflection characteristics obtained from both analyses again agree with experimental results, and correlation between results from both analytical models was very good. The symmetry of the horizontal structure is very well described by the analytical models. Table 3 summarizes the percentage error between the slopes of the test and analysis curves for load cycles 1 to 4.

The discrepancies between experimental and analytical results in some tests increase with increasing load and deflection. These discrepancies may be caused by the way the target-plate assembly is attached to the ribs. Before any loads are applied to the ribs or plate ends, the target plates are perpendicular to the proximity probes. When the ribs are displaced by the applied load, the target plates, which are fixed to the ribs, follow the rib displacement; the rib displacement includes rotation. In its final position, the target plate is at an angle with the proximity probe. Therefore, the measured vertical displacement is not the vertical component of the displacement vector of the point of interest on the rib. The error is a function of the horizontal displacement of the target-plate center and the angle the target plate makes with the horizontal. Some of the discrepancies between experimental and analytical results could have been eliminated if swivel joints were used to attach the target assembly to the ribs.

Results of Vibrational Analysis

Vibrational analysis of the reflector has been conducted to correlate results from both analytical models and for future correlation with experimental data. The first 13 natural frequencies for the reflector in its inclined position, obtained from large-displacements nonlinear analysis and small-displacements nonlinear analysis, are listed in table 4. Corresponding mode shapes are shown in figure 16 for the large-displacements nonlinear model. The eigenvalues and mode shapes obtained from the two analytical models show close agreement.

The first global mode shape identified, mode 4, exhibits a rocking motion of the reflector about the hub. Mode 9, the second global mode, involves torsion of the reflector around the hub center. Modes 1 to 3 and 6 to 8 are different combinations of first bending modes of the individual ribs. Second rib bending modes are in mode 10. Many of the mode shapes are similar and have similar frequencies because of the symmetry of the structure.

Frequency-response functions for random excitation at rib 2 were also generated by using the NASTRAN models. The plot in figure 17 shows a typical frequency-response function (FRF) taken in the vertical plane for rib 2. The point of excitation was the connection between the rib and sensor plate, and the measurement was taken 2.5 ft along the rib from the connector. The two analytical models show similar results.

Concluding Remarks

Two different nonlinear finite-element models for the final design of the Controls-Structures Interaction Evolutionary Model (CEM) reflector were developed and load-deflection data bases were generated for comparison with experimental results. Static tests to obtain load-deflection characteristics of the Controls-Structures Interaction (CSI) Evolutionary Model reflector were conducted. Limited vibrational analysis was also conducted, and preliminary system modes were computed for future system identification.

Excellent agreement between small-displacements and large-displacements nonlinear models for the reflector has been demonstrated. The modeling techniques described could be used in future applications involving the analysis of prestressed structures with complex geometry. The small-displacements nonlinear analysis approach works well for the analysis of prestressed structures where both the shape and the preload are known. During the design stage, the large-displacements nonlinear analysis approach can be used to design shape and prestress simultaneously.

Analytical and experimental results follow similar trends, but there are some discrepancies. These discrepancies may be reduced by modifying the displacement measurement hardware and by incorporating composite material data for the sensor plate into the finite-element models. Further refinement of the swivel-head bolt model is also warranted.

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Table 1. Axial-Force Distribution on Rib 7 Under Gravity and Target Weight for Large-Displacements Nonlinear Analysis

Rib beam element	Axial force on rib elements, lb, for CBAR connector	Axial force on rib elements, lb, for CROD connector
1	−11.87	−10.83
9	−12.66	−11.52
17	−13.12	−11.90
25	−13.50	−12.23
33	−13.65	−12.38
41	115.0	−9.77
49	115.0	−9.11

Table 2. Static Test Matrix for Inclined and Horizontal Positions of Reflector

Load cycle	Load location	Load range, lb	Increment, lb
1	All rib tips	0 to 2	0.5
2	Rib tips 5 and 7	0 to 1.5	0.5
3	All plate ends	0 to 24	3.0
4	Plate ends 1 and 3	0 to 24	3.0

Table 3. Test and Analysis Curve-Fitting Errors

Load cycle; measurement location	Small-displacements analysis versus large-displacements analysis, percent error	Test versus small-displacements analysis, percent error	Test versus large-displacements analysis, percent error
Inclined position			
1; Rib 1	8	10	16
1; Rib 3	6	33	25
1; Rib 5	29	11	14
1; Rib 7	2	18	16
2; Rib 1	13	36	17
2; Rib 3	13	22	7
2; Rib 5	5	12	6
2; Rib 7	3	20	16
Horizontal position			
3; Plate ends 1, 3, 5, and 7	2	14	12
4; Plate ends 1 and 3	2	8	9
4; Plate ends 5 and 7	2	19	17

Table 4. Analytical Natural Frequencies for Reflector

Mode	Frequencies, Hz, for small-displacements analysis	Frequencies, Hz, for large-displacements analysis
1	2.524	2.524
2	2.994	3.063
3	2.995	3.064
4	3.172	3.253
5	3.219	3.301
6	3.517	3.563
7	3.529	3.567
8	3.757	3.792
9	5.613	5.447
10	6.583	6.350
11	10.178	9.826
12	10.357	9.995
13	10.895	10.601

Appendix A

Listing of Finite-Element Analyses

L-92-14

Figure 1. Controls-Structures Interaction Evolutionary Model (CEM)

L-92-15

Figure 2. Controls-Structures Interaction Evolutionary Model reflector.

Figure 3. Side and top views of reflector. All linear dimensions are in inches.

L-92-16

Figure 4. Detailed view of connections.

Figure 5. Side and top views of reflector in horizontal position. All linear dimensions are in inches.

Figure 6. Side and top views of reflector in inclined position. All linear dimensions are in inches.

Figure 7. Typical truss strut and node-ball joint.

Figure 8. Composite panel cross-sectional element.

Figure 9. Rib analytical geometry for initial and prestressed states.

Figure 10. Large-displacements nonlinear analysis data-base dependent steps.

Figure 11. Small-displacements nonlinear analysis data-base independent steps.

Figure 12. Sensitivity of rib displacement under gravity and target weight loads to changes in swivel-head bolt model.

Figure 13. Load application and displacement measurement setup.

(a) Load cycle 1: symmetric loading of ribs.

(b) Load cycle 2: asymmetric loading of ribs.

Figure 14. Symmetric and asymmetric load-deflection characteristics of ribs. Inclined position.

(a) Load cycle 3: symmetric loading of plate ends.

(b) Load cycle 4: asymmetric loading of plate ends.

Figure 15. Symmetric and asymmetric loading of plate ends. Horizontal position.

(a) Mode 1; 2.54 Hz.

(b) Mode 2; 3.063 Hz.

Figure 16. Large-displacements analysis.

(c) Mode 3; 3.064 Hz.

(d) Mode 4; 3.253 Hz.

Figure 16. Continued.

(e) Mode 5; 3.301 Hz.

(f) Mode 6; 3.563 Hz.

Figure 16. Continued.

(g) Mode 7; 3.567 Hz.

(h) Mode 8; 3.792 Hz.

Figure 16. Continued.

(i) Mode 9; 5.447 Hz.

(j) Mode 10; 6.350 Hz.

Figure 16. Continued.

(k) Mode 11; 9.826 Hz.

(l) Mode 12; 9.995 Hz.

(m) Mode 13; 10.601 Hz.

Figure 16. Concluded.

Figure 17. Vertical frequency-response function for rib 2.